# 6 Multiple-Screw Pumps

Rotary screw pumps have existed for many years and are manufactured around the world. More demanding service requirements impose challenges on screw pump manufacturers to provide higher pressure or flow capability, better wear resistance, improved corrosion resistance, and lower leakage emissions. Better materials and more precise machining techniques as well as engineering innovation have led to improvements in all these areas.

The first screw pump built was probably an Archimedes design used to lift large volumes of water over small vertical distances; they are still manufactured and used for this service. Today's three-screw, high performance pump can deliver liquids to pressures above 4500 psi and flows to 3300 gpm with long-term reliability and excellent efficiency. Power levels to 1000 hp are available. Twin-screw pumps are available for flow rates to 18,000 gpm, pressures to 1450 psi, and can handle corrosive or easily stained materials, again at good efficiencies. Power ranges to 1500 hp operate on critical applications.

In multiple-screw pumps, each wrap of screw thread effectively forms a stage of pressure capability. High pressure pumps have 5 to 12 stages or wraps where low pressure pumps may have only 2 or 3 wraps. This staged pressure capability is illustrated in Figure 30. More wraps are incorporated in pumps designed for higher pressure service.

## THREE-SCREW PUMPS

Three screw pumps are the largest class of multiple-screw pumps in service today.[10] They are commonly used for machinery lubrication, hydraulic elevators, fuel oil transport and burner service, powering hydraulic machinery and in refinery processes for high temperature viscous products such as asphalt, vacuum tower bottoms, and residual fuel oils. Three-screw pumps, also are used extensively in crude oil pipeline service as well as the gathering, boosting, and loading of barges and ships. They are common in engine rooms on most of the world's commercial marine vessels and many combat ships. Subject to material selection limitations, three-screw pumps are also used for polymer pumping in the manufacture of synthetic fibers such as nylon and lycra. Designs are available in sealless configurations such as magnetic drives and canned arrangements (see Figure 31).

The magnetic drive screw pump is used extensively for pumping isocyanate, a plastic component which is an extremely difficult fluid to seal using conventional

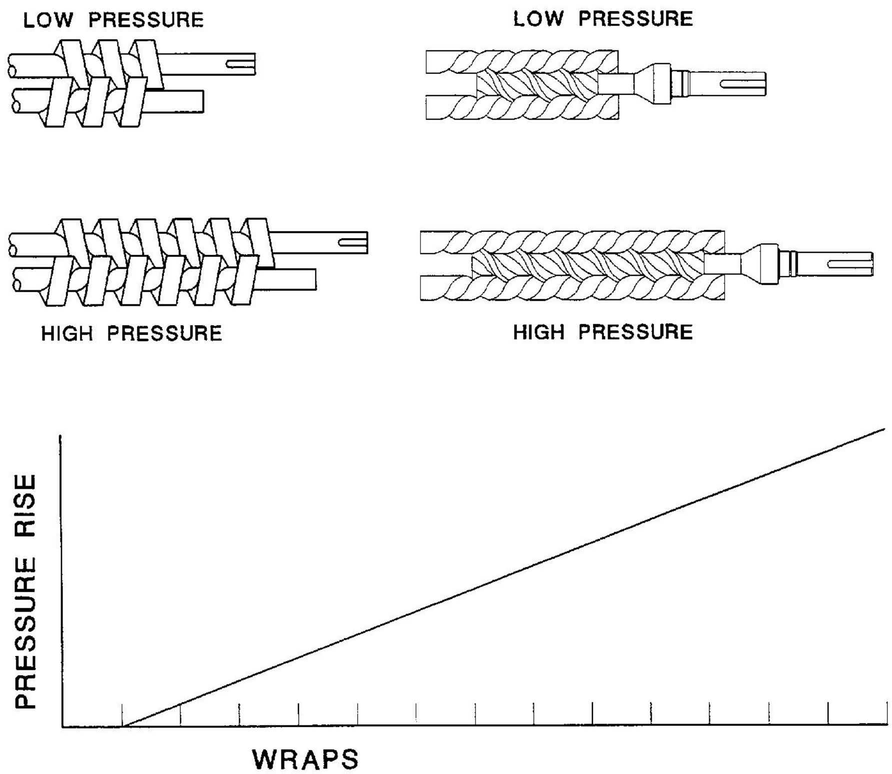


**FIGURE 30** Staging effects of multiple-screw pumps. (Courtesy of IMO Pump.)

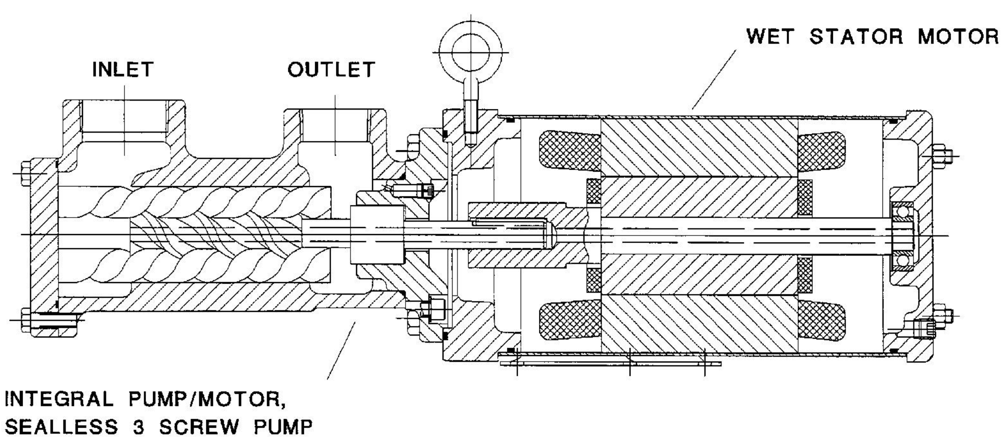


**FIGURE 31** Three-screw canned motor pump. (Courtesy of IMO Pump.)

technology. Three-screw pumps are renowned for their low noise levels, high reliability, and long life. They are not, however, very low cost pumps.

## Design and Operation

Three-screw pumps are manufactured in two basic styles: single suction and double suction (see Figure 32).

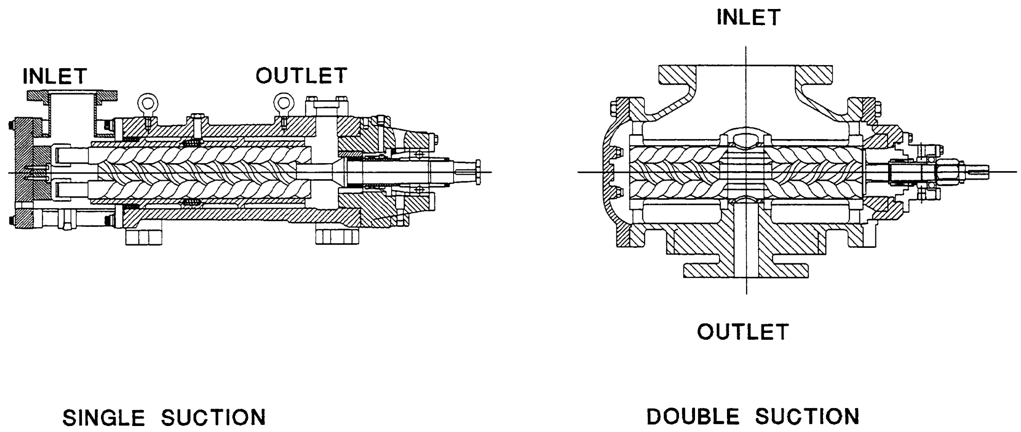


**FIGURE 32** Single and double suctions three-screw pumps. (Courtesy of IMO Pump.)

The single suction design is used for low to medium flow rates and low to very high pressure. The double suction design is really two pumps parallel in one casing. They are used for medium to high flow rates at low to medium pressure.

Three-screw pumps generally have only one mechanical shaft seal and one, or perhaps two, bearings that locate the shaft axially. Internal hydraulic balance is such that axial and radial hydraulic forces are opposed and cancel each other. Bearing loads are thus very low. Another common characteristic of three-screw pumps is that all but the smallest low pressure designs incorporate replaceable liners in which the pumping screws rotate. This simplifies field repairs.

The center screw, called the power rotor, performs all the pumping. The meshed outside screws, called idler rotors, cause each liquid-holding chamber to be separated from the adjacent one except for running clearances. This effectively allows staging of the pump pressure to rise. Because the center screw is performing all the pumping work, the drive torque transferred to the idler rotors is only necessary to overcome the viscous drag of the cylindrical rotor spinning within its liner clearance. The theoretical flow rate of these pumps is a function of speed, screw set diameter, and the lead angle of the threads. Flow rate is a function of the cube of the center screw diameter. Slip, however, depends on clearances, differential pressure, and viscosity, and is only a square of the power rotor diameter. This results in larger pumps being inherently more efficient than smaller pumps, a fact that applies to most rotating machinery.

Speed is ultimately limited by the application's capability to deliver flow to the pump inlet at a sufficient pressure to avoid cavitation. This is true of all pumps. Three-screw pumps tend to be high speed pumps, not unlike centrifugal pumps. Two-pole (3600 RPM) and four-pole motors are most commonly used. When large flows, very high viscosities, or low available inlet pressures dictate, slower speed may be necessary. For example, some polymer services handle liquid at 250,000 SSU or more. Three-screw pumps on such service would typically be operated in the 50 to 150 RPM range. Gas turbine fuel injection service would more commonly be in the 3000 to 3600 RPM range since the fuels tend to be low in viscosity (1 to 20 centistokes), and the pump inlet is normally boosted to a positive pressure from a fuel treatment skid pump. The high speed operation is desirable when handling low viscosity liquids since the idler rotors generate a hydrodynamic liquid film in their load zones that resists radial hydraulic loads, very similar to hydrodynamic sleeve bearings found in turbomachinery.

In order to achieve the highest pressure capability from three-screw pumps, it is necessary to control the shape of the screws while under hydraulic load. This is best achieved by the use of five axis NC profile grinding which allows complete dimensional control and a high degree of repeatability. Opposed loading of the idler rotor outside diameters on the power rotor root diameter dictate that these surfaces be heat treated to withstand the cyclic stresses. Profile thread grinding allows the final screw contour to be produced while leaving the rotors quite hard, on the order of 58 RC (58 on the Rockwell C scale). This hard surface better resists abrasive wear from contaminants.

Because some three-screw pump applications range to pressures of 4500 psi, pumping element loading due to hydrostatic pressure can be quite high. Without hydraulic balance to counteract this loading in one or two planes, bearing loads would be excessive and operating life shortened.

Single-ended pumps use two similar, but different, techniques to accomplish axial hydraulic balance. The center screw, called a power rotor, incorporates a balancing piston at the discharge end of the screw thread (see Figure 33a). The area of the piston is made about equal to the area of the power rotor thread exposed to discharge pressure. Consequently, equal opposing forces produce zero net axial force due to discharge pressure, and place the power rotor in tension. The balance piston rotates within a close clearance stationary bushing, which may be hardened or hard coated to resist erosive wear. The drive shaft side of the piston is normally internally or externally ported to the pump inlet chamber. Balance leakage flow across this running clearance flushes the pump mechanical seal which remains at nominal pump inlet pressure. The two outer screws, called idler rotors, also have their discharge ends exposed to discharge pressure. Through various arrangements, discharge pressure is introduced into a hydrostatic pocket area at the inlet end of the idler rotors (see Figure 33b).

The effective area is just slightly less than the exposed discharge end area, resulting in approximately equal opposing axial forces on the idler rotors. The idler rotors are therefore in compression. Should any force cause the idler rotor to move toward discharge, a resulting loss of pressure, acting on the cup shoulder area or hydrostatic land area, tends to restore the idler rotor to its design running position. The upper view in Figure 33b shows a stationary thrust block (cross hatched) and a stationary, radially self-locating balance cup. Discharge pressure is brought into the cup via internal passages within the pump or rotor itself. The lower view shows a hydrostatic pocket machined into the end face of the idler rotor. It too is fed with discharge pressure. The gap shown is exaggerated and is actually very near zero.

For some contaminated liquid services, the hydrostatic end faces of the idler rotors are gas-nitride-hardened or manufactured from solid tungsten carbide, and shrink fitted to the inlet end of the idler rotors. When the cup design is used, the cup inside the diameter and shoulder area are normally gas–nitride hardened. Both techniques are used to resist wear due to the fine contaminants.

(a) piston

(b) hydrostatic packet

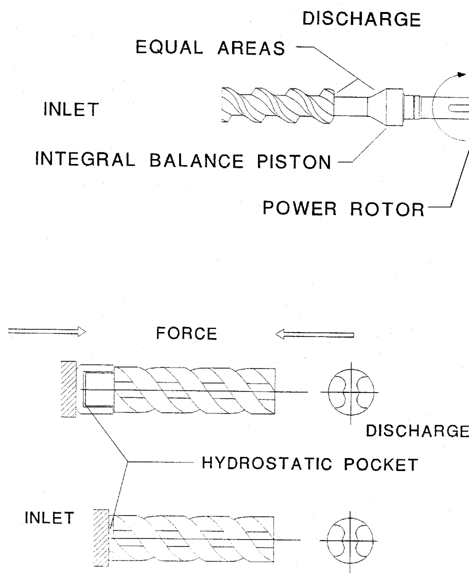


**FIGURE 33** Hydraulic thrust balancing. (Courtesy of IMO Pump.)

In a radial direction, three-screw pumps achieve power rotor hydraulic balance due to symmetry. Equal pressure acting in all directions within a stage or wrap results in no radial hydraulic forces, since there are no unbalanced areas. The power rotor will frequently have a ball bearing to limit end float for proper mechanical seal operation, but it is otherwise under negligible load. Idler rotor radial balance is accomplished through the generation of a hydrodynamic liquid film, and operates similar to a journal or sleeve bearing.

The eccentricity of the rotating idler rotor sweeps liquid into a converging clearance resulting in a pressurized liquid film. The film pressure acts on the idler rotor's outside diameters in a direction opposing the hydraulically generated radial load. Increasing viscosity causes more fluid to be dragged into the pressurized film, causing the film thickness and pressure supporting capability to increase. The idler rotors are supported in their respective housing bores on liquid films and have no other bearing support system. Within limits, if differential pressure increases, the idler rotor moves radially toward the surrounding housing bores. The resulting increase in eccentricity increases the film pressure and maintains radial balance of the idler rotors.

In three-screw pumps, inlet pressure above or below atmospheric pressure will produce an axial hydraulic force on the drive shaft. In most pump applications, pump inlet pressures are below, or slightly above, atmospheric pressure so the forces generated by this low pressure acting on a small area are negligible. However, if the application requires the pump to operate at an elevated inlet pressure, from a booster pump for example, then inlet pressure acting on the inlet end of the power rotor is only partly balanced by this same pressure acting on the shaft side of the balance piston. In effect, the area of the power rotor at the shaft seal diameter is an unbalanced area. This area multiplied by the inlet pressure is the resulting axial load, toward the shaft end. When specifying pumps, it is important to clearly state the maximum expected inlet pressure so the pump manufacturer can verify this loading. Several reliable methods are in use to carry this load, including antifriction bearing such as double extending the power rotor out the inlet end of the pump (which adds a second shaft seal) or sizing the balance piston to counterbalance this axial force.

As versatile as three-screw pumps are, they are not suitable for some applications. While many advances in materials engineering are taking place, the state of the art for three-screw pumps is such that very corrosion-resistant materials, such as high nickel steels, have too great a galling tendency. The rotors of three-screw pumps touch, and thus any materials that tend to gall are unsuitable. Unfortunately, this includes many corrosion-resistant materials. Viscosities too low to allow hydrodynamic film support generation are also application areas for which the three-screw pump is not optimal.

## TWO-SCREW PUMPS

Generally, two-screw (or twin-screw) pumps are more costly to produce than three-screw pumps and thus are not in as extensive use. They can, however, handle applications that are well beyond many other types of pumps, including three-screw designs. Twin-screw pumps are especially suited to very low available inlet pressure applications, and more so if the required flow rates are high. Services similar to three-screw pumps include: crude oil pipelining; refinery hot, viscous product processing, synthetic fiber processing; barge unloading; fuel oil burner and transfer; as well as unique applications such as: adhesive manufacture; nitrocellulose explosive processing; high water cut crude oil; multiphase (gas/oil mixtures) pumping; light oil flush of hot process pumping; cargo off-loading with ballast water as one of the fluids; and tank stripping service where air content can be high and paper pulp production needs to pump over about 10% solids.

### Design and Operations

The vast majority of twin-screw pumps are of the double suction design (see Figure 34). The opposed thread arrangement provides inherent axial hydraulic balance due to symmetry. The pumping screws do not touch each other and thus lend themselves well to manufacture from corrosion-resistant materials. The timing gears serve to both synchronize the screw mesh as well as to transmit half the total power input from the drive shaft to the driven shaft. Each shaft effectively handles half the flow, and thus half the power. Each end of each shaft has a support bearing for the unbalanced radial hydraulic loads. A few designs leave the bearings and timing gears operating in the liquid pumped. While this results in a significantly lower cost pump design, it defeats much of the value that twin-screw pumps bring to applications. The more common and better design keeps the timing gears and bearings external to the liquid pumped. They need not rely upon the lubricating qualities of the pumped liquid or its cleanliness. Four mechanical shaft seals keep these bearings and timing gears isolated and operating in a controlled environment.

Hydraulic radial forces on a two-screw pump rotor due to differential pressure are illustrated in Figure 35. The forces are uniform along the length of the pumping threads. These hydraulic forces cause deflection “y” for which running clearance

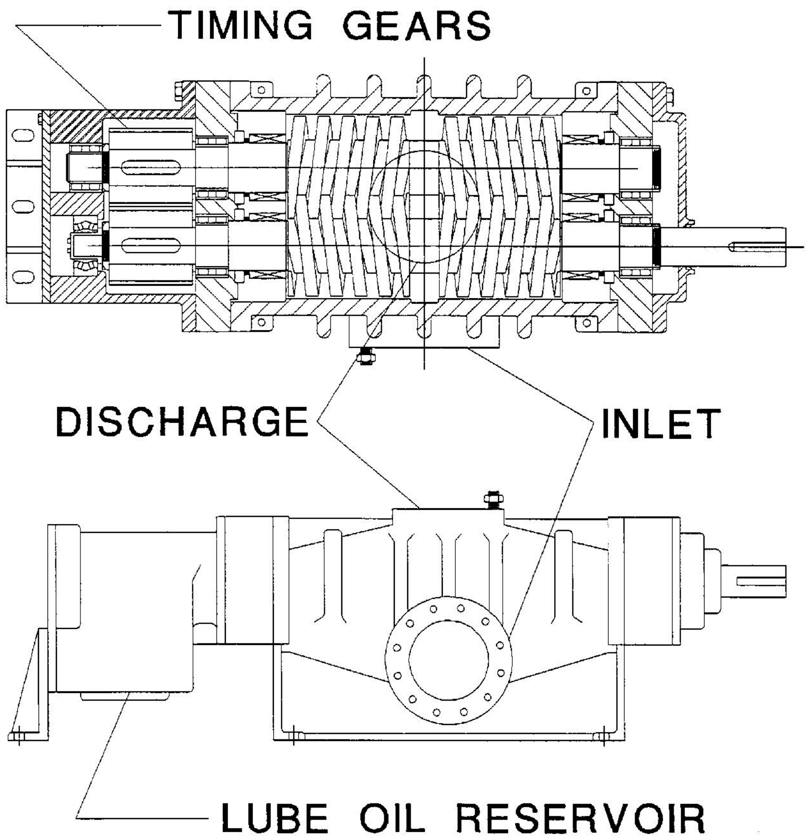


**FIGURE 34** Double suction two-screw pump. (Courtesy of IMO Pump.)

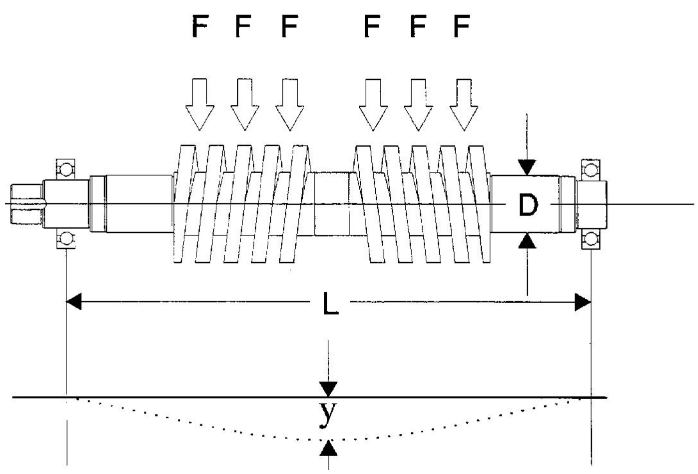


**FIGURE 35** Radial forces in twin-screw pumps. (Courtesy of IMO Pump.)

must be provided in the surrounding pump body. Greater deflection requires larger clearances, resulting in more slip flow or volumetric inefficiency, so "y" must be kept to a minimum. Excessive deflection will cause damage to the surrounding body and/or contribute to rotating bend fatigue which will ultimately result in shaft breakage. The following is the general form of the deflection equation,

$$y = \frac{F \times L^3}{c \times E \times I} < \text{radial clearance} \quad (39)$$

where "F" is the summation of the hydraulic forces, "L" is the bearing span, "c" is a constant, "E" is the shaft material modulus of elasticity, and "I" is the shaft moment of inertia. The shaft moment of inertia is a function of "$D^4$," where "D" is the effective shaft diameter. This equation is simplified and, in practice, must account for the varying shaft and screw diameters as they change along the length of the rotor. If screw "shells" are not integral with the shaft, that is, not made from a single piece of material, then material differences as well as attachment schemes must be factored into the deflection calculation. In any event, it is easy to see that the bearing span, "L," must be kept to a minimum to minimize deflection. The use of large diameter shafts and screw root sections helps to maintain minimum deflection.

Depending on the direction in which the threads are machined (left or right hand), and the direction of shaft rotation, the pump manufacturer can predetermine the deflection to be in either of the two radial directions: up or down, for a horizontal pump. These radial deflection loads are absorbed through externally lubricated anti-friction bearings. Higher differential pressure produces higher radial loads or forces. Smaller lead angles of the screw set reduce both these radial loads and the flow rate. Larger lead angles increase both flow rate and radial loading. Bearings are usually sized to provide 25,000 or more hours, "L10,"[8] bearing life at the maximum allowable radial loading and maximum design operating speed. Because of this pumpage-independent bearing system, two-screw pumps with external timing gears and bearings can handle high gas content as well as light oil flushes, water, etc.

Twin-screw pumps are manufactured from a broader range of materials, including 316 stainless steel. When extreme galling tendencies exist between adjacent running components, a slight increase in clearance is provided to minimize potential for contact. In addition, the stationary bores in which the screws rotate can be provided with a thick industrial hard chrome coating which further reduces the likelihood of galling as well as providing a very hard, durable surface for wear resistance. Such coatings do, however, require the capability of inside diameter grinding to achieve finished geometry within tolerances. For highly abrasive services, the outside diameter of the screws can be coated with various hard facings to better resist wear. Among these coatings are tungsten carbide, stellite, chrome oxide, alumina titanium dioxide, and others. Figure 36 shows a finished screw with a hard surfaced outside diameter. The high efficiency performance is a clear advantage over centrifugal pumps where liquid viscosity exceeds 100 SSU (20 centistokes). For pressures requiring two or more stages in a centrifugal pump, multiple-screw pumps frequently will be very competitive on a first cost basis as well. Operating liquid temperatures as high as 600°F have been achieved in twin-screw pumps for the ROSE® deasphalting process (see Figure 37). Timing gears and bearings are force cooled while the pump body is jacketed for a hot oil circulating system to bring the pump to process temperature in a gradual, controlled manner. Three-screw pumps have been applied to the same elevated temperature, more commonly on asphalt or vacuum tower bottoms services in refineries.

**FIGURE 36** Twin-screw thread shell with hard coated O.D. (Courtesy of IMO Pump.)

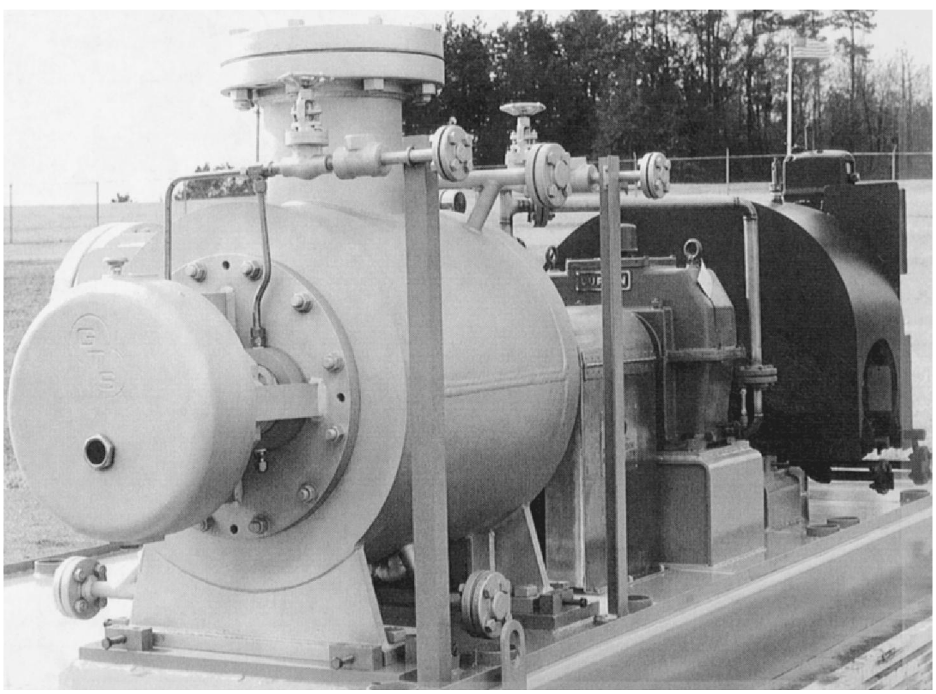

**FIGURE 37** Rose® process 600°F twin-screw pump. (Courtesy of IMO Pump.)

Medium and high viscosity operation are not the only regions where multiple-screw pumps bring advantages to the end user. Low viscosity, combined with high pressure and flows less than approximately 450 gpm, are excellent screw pump applications. Continuous, non-pulsating flow is required, for example, in high pressure atomizers for fuel combustion. Combustion gas turbines frequently burn distillate fuels, naphtha, and other low viscosity petroleum liquids that may reach

1 centistoke or less and require pumping pressures in the 950 to 1300 psi range. The combination of modest flow, low viscosity, and high pressure is a difficult service for all but reciprocating pumps. The pressure and flow pulsation from reciprocating pumps usually cannot be tolerated in fuel burning systems, especially combustion gas turbines.

Ongoing research and development efforts will further extend the capabilities of these machines allowing better performance over a broader range of applications. Multiple-screw pumps are uniquely suited to many of the applications described herein and offer long-term benefits to their users.

## USER COMMENTS

The following comments on various types of pumps were made by the chemical plant users interviewed:

> Multiple-screw pumps are the most technically complex and expensive of all rotary pumps. However, they have certain very important advantages. Applied to high pressures (2000 psi range) and high temperatures (designs are known to 800°F range), they can operate at a wide range of viscosities, from grease to rubber. Due to close clearance, these pumps have high efficiencies. Timing gears may, or may not, be required, depending on the design (two- vs. three-screw designs). Pinned rotors could be prone to failures but are a less expensive option. With external bearings and timing gears, four seals are required, and field work, for alterations and repair, is not trivial. Advantages of screw pumps are their quiet operation, low pulsations, and good NPSHR (net positive static suction head required) characteristics.